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TURBOMACHINERY CHARACTERISTICS OF BRAYTON CYCLE SPACE-POWER GENERATION SYSTEMS

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ABSTRACT

This paper presents the results of an analytical study of turbomachinery requirements and configurations for Brayton cycle space-power systems. Basic turbomachinery requirements are defined and typical effects of such system design parameters as power, temperature, pressure, and working fluid on turbomachinery geometry and performance are explored. Typical turbomachinery configurations are then presented for systems with power outputs of 10, 100, and 1000 KW.

AUTHOR

INTRODUCTION

Electric power levels for space missions are continually increasing as the complexity and the duration of these missions increase. The most promising power-generation technique for near-future application to missions requiring power levels in excess of several kilowatts for extended periods of time appears to be the closed-loop heat engine. Such a system utilizes the conversion, by means of a thermodynamic cycle, of nuclear or solar heat to mechanical shaft power which, in turn, is converted to electrical power by means of an alternator. The two thermodynamic cycles most often considered for this application are the Rankine cycle, utilizing a boiling and condensing metal working fluid, and the Brayton cycle, utilizing a gaseous working fluid.

Brayton cycles (open loop) have performed most satisfactorily in turboprop and turbojet engines but have not been as seriously considered for space-power systems as Rankine cycles have been. The ability of the Rankine cycle to reject heat at a constant temperature results in radiator areas and weights smaller than those for a Brayton cycle. For certain applications, however, where low powerplant weight is not a critical requirement, the Brayton cycle merits strong consideration because it has many features that give it the potential of being an extremely reliable system. These features include (b) the single phase flow throughout the system, (a) the use of a noncorrosive working fluid, and (c) the absence of erosion and cavitation in the turbomachinery. In addition, much of the basic technology for the Brayton cycle, and especially for the turbomachinery components, is presently available through the jet engine and gas turbine power production fields.

In view of these considerations, an investigation of Brayton cycle space-power systems was undertaken at the Lewis Research Center. Studies were made of the thermodynamic characteristics of such systems (1)[1] as well as of the heat-transfer components (2) and the turbomachinery components. A discussion of system weight and reliability characteristics is presented in reference 3. It is the purpose of this paper to define the basic turbomachinery requirements for a Brayton cycle space-power system, and then to discuss and to show how the various system design parameters interact with these requirements and the resultant effects on turbomachinery geometry.

[1]Numbers in parenthesis refer to similarly numbered references in bibliography at end of paper.

NOMENCLATURE

A	flow area, sq ft
c_p	heat capacity, Btu/(lb)($^{\circ}R$)
D	diffusion factor
d	diameter, ft
g	gravitational constant, 32.2 ft/sec^2
H	specific work, ft-lb/lb
Δh	specific work, Btu/lb
J	mechanical equivalent of heat, 778 ft-lb/Btu
M	molecular weight
N	rotative speed, rpm
N_R	Reynolds number
N_s	specific speed
n	number of stages
P	system-power output, kw
p	absolute pressure, lb/sq in.
Q	volume-flow rate, cu ft/sec
R	universal gas constant
r	radius, ft
s	stress, lb/sq in.
T	absolute temperature, $^{\circ}R$
U	blade speed, ft/sec
V	absolute gas velocity, ft/sec
w	weight-flow rate, lb/sec
y_a/y_r	ratio of disk thickness at axis to disk thickness below rim

γ	specific heat ratio
η	efficiency
λ	speed-work parameter
μ	gas viscosity, lb/(ft)(sec)
ν	blade-jet speed ratio
ρ	density, lb/cu ft
σ	blade solidity

Subscripts:

b	blade
c	compressor
d	disk
ex	exit
h	hub
id	ideal
m	blade mean section
r	rim
s	static
T	turbine
t	tip
u	tangential component
x	axial component
1	compressor inlet
2	compressor exit
3	heat source inlet
4	turbine inlet

5 turbine exit

6 radiator inlet

Superscript:

- overall

CYCLE CHARACTERISTICS

Cycle Description

A schematic diagram of a Brayton cycle system is shown in Fig. 1(a) and the corresponding temperature-entropy diagram in Fig. 1(b). The radiator exit gas at point 1 is compressed to point 2 and then passes through the recuperator where it is heated to point 3 by the hotter gas from the turbine. Final heating of the gas to its maximum temperature, point 4, takes place in the heat source, which can be either a solar absorber or nuclear reactor. The hot gas then expands through the turbine to point 5, thereby producing the mechanical work necessary to drive the compressor and alternator. From the turbine the gas passes through the recuperator where it is cooled to point 6 as it transfers heat to the compressor exit gas. Final cooling of the gas to point 1 takes place in the radiator, where the excess heat is rejected to space. Alternate configurations can include a liquid heating loop and/or a liquid cooling loop (2). The gas loop, however, remains the same as described previously except that heat exchangers replace the heat source and/or radiator.

Cycle Thermodynamics

The thermodynamic characteristics of Brayton cycle space-power systems were presented in reference 1. Since the radiator is a major size and weight contributor to the powerplant, minimum radiator area was chosen as the logical criterion for initial selection of the cycle variables. This criterion, as explained in reference 1, can be used for solar as well as nuclear systems. The thermodynamic analysis shows that (1) the requirement for minimum radiator area determines the range of turbomachinery pressure ratios to be considered, and (2) the turbomachinery performance significantly affects system performance and size.

The manner in which pressure ratio is determined can be seen from Fig. 2, where relative radiator area for a typical system is plotted against the ratio of compressor inlet temperature to turbine inlet temperature (T_1/T_4) for several values of turbine temperature ratio (T_5/T_4). Compressor pressure ratios corresponding to the turbine temperature ratios are also shown in the figure. A cross-plot of the data in Fig. 2 would show that in order to obtain radiator areas within about 10 percent of the minimum area for this particular case, compressor pressure ratios of between about 2.2 and 4 are required. These pressure ratio limits will vary somewhat from system to system, but the range is generally from about 2 to 4. For the case shown in Fig. 2, the assumed system pressure losses are such that the turbine pressure ratios are 85 percent of those for the compressor. The ramifications of the selected pressure ratios on the turbomachinery characteristics will be discussed in a subsequent section.

System performance and size depend to a great extent on turbomachinery performance. This can be seen from Fig. 3, where relative radiator area and relative cycle efficiency are plotted against turbine efficiency for several values of compressor efficiency. The relative areas represent the minimum achievable area (as determined from plots similar to Fig. 2) for each combination of turbine and compressor efficiencies, and the relative cycle efficiencies are those for these minimum area points. On the average, a 5-point decrease in turbine efficiency results in a 30- to 35-percent increase in radiator area, while a 5-point decrease in compressor efficiency results in a 25- to 30-percent increase in radiator area. The corresponding cycle efficiencies decrease by about 5 to 10 percent for a 5-point decrease in either turbine or compressor efficiency. Since the radiator areas and weights for Brayton cycle space-power systems are inherently quite large, the achievement of high turbomachinery performance is a prime design requirement.

TURBOMACHINERY REQUIREMENTS

There are two basic requirements that Brayton cycle turbomachinery must meet. As shown in the previous section, good performance is one prime requirement. Since space-power systems will be expected to operate for periods of time greater than 1 year, high reliability is another prime requirement. These two requirements and the methods for achieving them are now discussed.

Performance

The aerodynamic design of turbines and compressors, in general, can be divided into three phases. The first phase is the selection of the overall requirements such as fluid, temperature and pressure levels, flow rate, power, and perhaps a required efficiency level. Determination of the machine geometry that best meets these requirements is the second phase. This phase is generally executed through the calculation of free-stream velocity diagrams and results in the determination of the number of stages, machine and blade sizes, and blade turning. The third phase is then the evolution of the blade shapes that will meet the velocity-diagram requirements.

The expected efficiency of the machine that is evolved to meet the requirements set up in the first phase depends on both the overall geometry selected in phase two and the detailed blading design in phase three. Efficiency is a function of the second phase through the aerodynamic severity of the selected velocity diagrams and a function of the third phase through such blade and mechanical design features as surface loading, trailing-edge blockages, and tip and running clearances. For this general study, detailed blade designs were not evolved. Assuming that efficient blade and mechanical designs can be accomplished, the level of efficiency becomes primarily a function of the selected velocity diagrams.

A number of basic parameters are available for relating efficiency to the velocity diagrams. Certain of these parameters were derived for one particular type of machine and, consequently, are used only for that type. Axial-flow turbines are generally designed at NASA using the stage speed-work parameter, λ, and/or the stage blade-jet speed ratio, ν (4). A satisfactory criterion for designing axial-flow compressors at NASA has been the diffusion factor, D (5). A criterion that is generally applicable to all types of turbomachinery and which was used for the radial-flow turbines and compressors in this study is specific speed, N_s(6,7). These parameters will now be discussed individually.

Axial-flow turbines. - The speed-work parameter, λ, was used as the basic parameter for relating the efficiency of axial-flow turbines to the velocity-diagram quantities. For a stage, this parameter is defined as

$$\lambda = \frac{U_m^2}{gJ\,\Delta h}$$

and can be considered as the ratio of the energy associated with the mean-section blade speed, U_m, to the stage specific work output, Δh. Since the stage specific work output can be written in terms of velocity-diagram quantities as

$$\Delta h = \frac{U_m\,\Delta V_u}{gJ}$$

another expression for λ is obtained as

$$\lambda = \frac{U_m}{\Delta V_u}$$

This expression shows that λ is a measure of the severity of the stage velocity diagram in terms of the relation of the blade speed to the change in tangential velocity, ΔV_u. A detailed discussion of the types of velocity diagrams associated with various values of λ and certain limitations imposed on these velocity diagrams is given by Stewart (4). For the general case of a multistage turbine with constant stage work and constant mean-section blade speed, an overall speed work parameter, $\bar{\lambda}$, can be defined and related to λ as

$$\bar{\lambda} = \frac{U_m^2}{gJ\,\overline{\Delta h}} = \frac{U_m^2}{ngJ\,\Delta h} = \frac{\lambda}{n}$$

Another parameter often used in describing turbine efficiency is the blade-jet speed ratio, $\bar{\nu}$. This parameter is defined as the ratio of the mean-section blade speed to the jet velocity corresponding to the ideal expansion from inlet total to exit static conditions across the turbine. That is,

$$\bar{\nu} = \frac{U_m}{\sqrt{2gJ\,\overline{\Delta h}_{id,s}}}$$

A relation between $\bar{\nu}$ and $\bar{\lambda}$ can be obtained as

$$\bar{\nu} = \left(\frac{\bar{\eta}_s\bar{\lambda}}{2}\right)^{1/2}$$

This relation shows that the two parameters are related by the overall static efficiency $\bar{\eta}_s$; therefore, if efficiency is a function of one of these parameters, it must also be a function of the other.

A method for relating overall static efficiency to $\bar{\lambda}$, and consequently to $\bar{\nu}$, is derived by Stewart (4). A typical set of turbine efficiency characteristics, computed by this method, as a function of blade-jet speed ratio, $\bar{\nu}$, and number of stages is shown in Fig. 4. It is through the use of such curves that high-performance axial-flow turbines can be designed.

Axial-flow compressors. - In the analysis of axial-flow compressors, a principal problem is to establish a velocity diagram shape that gives a good balance between losses due to small blading and inefficiency due to higher-than-optimum inlet-flow angles. It was determined for this study that an inlet-flow angle of approximately 65 degrees at the mean radius gave a satisfactory compromise for a generalized analysis.

An aerodynamic stage loading limitation was used to determine the required number of stages. Of the several loading criteria that have been used for designing compressors, the diffusion factor, D, as developed by Lieblein, Schwenk, and Broderick (5) was chosen for this analysis. As given in reference 5, the diffusion factor can be expressed as

$$D = 1 - \frac{W_o}{W_i} + \frac{\Delta W_u}{2\sigma W_i}$$

where σ is the blade solidity, W_i and W_o are the relative velocities in and out of the blade row, and ΔW_u is the change in tangential component of relative velocity. Application of the diffusion factor to a large number of axial-flow compressors indicated that losses increase rapidly as D increases above 0.4 in the blade tip region and 0.6 at all other portions of the blade. In order to obtain maximum performance for axial-flow compressors, these limits were utilized in this study.

<u>Radial-flow machines.</u> - One of the parameters commonly used for correlating turbomachinery efficiency is specific speed, N_s, which is expressed as

$$N_s = \frac{NQ^{1/2}}{H_{id}^{3/4}}$$

where N is the rotative speed in rpm, Q is the low-pressure-side volume flow in cu ft/sec and H_{id} is the ideal specific work, in ft-lb/lb, based on total conditions across the stage. A certain value of N_s expresses all values of the variables N, Q, and H_{id} that lead to similar flow conditions in geometrically similar turbomachines. In order for N_s to be used as a practical parameter, the restriction is introduced that it is evaluated at the point of best efficiency. When this is done, N_s becomes a parameter of great significance because each different class of machine has its maximum efficiency within a relatively restricted range of specific speed, this range being different for each class.

Correlations for best efficiency point as a function of specific speed are shown in Fig. 5 for both radial-flow turbines and radial-flow compressors. The turbine curve is reproduced from Wood (6), while the compressor curve is based on the data of Samaras and Tyler (8) but was revised to reflect current technology, an example of which is presented by Senoo (9). Both curves were established from available test data. These relationships were used for the preliminary design of the radial-flow turbomachines considered in this study.

As mentioned at the beginning of this section, turbomachinery efficiency is not only a function of the parameters just discussed, but it also depends on blade and mechanical design features. In addition, Reynolds number and Mach number levels affect the achievable efficiency. The selection of optimum values for the previously discussed parameters, therefore, is necessary for, but does not guarantee, the achievement of maximum efficiencies. The effects of these other factors on efficiency are largely dependent on the selected system design parameters and will be subsequently discussed in conjunction with these design parameters.

Reliability

The use of an inert gas as the system working fluid, as pointed out in reference 3, results in a very attractive reliability potential for the Brayton cycle. Turbomachinery potential problem areas such as corrosion, erosion, and cavitation do not exist for this system. One major problem area facing Brayton cycle turbomachinery is the development of suitable gas bearings. The gas bearing, once properly developed, is potentially a long-life trouble-free mechanism. Operational tests for gas turbomachinery are relatively simple to conduct, so many tests of this type can be made. Problems of extended operation are principally related to fatigue and creep, and these primarily affect the turbine, which must operate in the area of maximum cycle temperature. Endurance can be designed into the machine through the use of allowable stress levels determined from rupture and creep data, and few extended tests are required. High reliability, consequently, can be demonstrated readily through component testing. Since the allowable stress is primarily a function of temperature, the ramifications of stress limitations on the turbomachinery will be discussed in a subsequent section dealing with the effect of temperature selection.

SYSTEM DESIGN PARAMETERS

System design parameters such as pressure ratio, fluid molecular weight, temperature, pressure, and power all affect significantly the turbomachinery geometry and performance. In many cases, certain limitations must be placed on the choice of these parameters in order that the basic requirements of high efficiency and reliability be achieved. It is the purpose of this section to discuss the effects of the design parameters on the geometry and the potential performance of the turbomachinery. Let it be noted that the figures to be presented in this discussion are for the purpose of showing typical trends and effects and are not meant to represent optimum design parameters for any particular application. For a given application, the optimum design parameters are determined from more detailed analyses.

Pressure Ratio

Pressure ratio limits, as mentioned previously, are set initially by the requirement for near minimum radiator area. For highly recuperative Brayton cycle systems, such as that represented by Fig. 2, radiator areas within 10 percent of the achievable minimum can be obtained with compressor pressure ratios of about 2 to 4. For nonrecuperative systems, the corresponding pressure ratios are about 2.5 to 4.5.

A typical effect of pressure ratio on efficiency, as given in reference 9, for a radial-flow compressor is shown in Fig. 6. For this example case, an increase in pressure ratio from 2 to 3.3 results in a decrease in achievable efficiency from 85 to 80 percent. This behavior results from the increased amount of required diffusion as well as the increased impeller loading. Such a decrease in compressor efficiency, as seen from Fig. 3, results in a significant increase in radiator area and a reduction in cycle efficiency. For an axial-flow compressor, an increase in pressure ratio results in an increase in the number of stages if a given efficiency is to be maintained. Since the required number of stages is relatively large, as will subsequently be shown, further increases should be avoided, if possible. The number of stages required for an axial-flow turbine is considerably less than for the axial-flow compressor; consequently, the choice of pressure ratio does not affect the turbine to the same extent as it does the compressor. As seen from this discussion, the pressure ratio should be maintained at or near the lower end of the range specified by radiator area considerations.

Working Fluid

For this study the candidate working fluids were limited to the monatomic inert gases in order to eliminate the possibility of chemical interaction of the working fluid with the structural materials. The gases considered include helium (M = 4), neon (M = 20), argon (M = 40), and krypton (M = 84). The choice of working fluid affects the turbomachinery primarily in two ways - (1) the number of stages required for the attainment of high efficiency and (2) the Reynolds number. In addition, there may or may not be a size effect, but this is of a secondary nature.

Number of stages. - The manner in which fluid molecular weight affects the required number of stages can be shown on the basis of the previously mentioned velocity parameters and their defining equations. As an example, the case of an axial-flow turbine with constant stage work and constant mean-section blade speed will be considered. For this case, the speed-work parameter, λ, can be defined as

$$\lambda = \frac{nU_m^2}{gJ\,\overline{\Delta h}} \tag{1}$$

Experience has shown that maximum efficiency is obtained with a λ of about 1. Therefore, setting $\lambda = 1$ and rearranging Equation (1) yields

$$n = \frac{gJ\,\overline{\Delta h}}{U_m^2} \tag{2}$$

Total work can be expressed as

$$\Delta h = c_p\,\Delta T = \frac{\gamma}{\gamma - 1}\,\frac{R}{JM}\,\Delta T = \frac{\gamma}{\gamma - 1}\,\frac{R}{JM}\,T_4\left[1 - \frac{T_5}{T_4}\right] \tag{3}$$

Substituting Equation (3) into Equation (2) yields

$$n = \frac{g\gamma RT_4\left(1 - \frac{T_5}{T_4}\right)}{(\gamma - 1)MU_m^2} \tag{4}$$

Expressing mean-section speed in terms of tip speed yields

$$U_m = \frac{U_t}{r_t} r_m = \frac{U_t}{r_t}\left(\frac{r_h + r_t}{2}\right) = \frac{U_t}{2}\left(\frac{r_h}{r_t} + 1\right) \tag{5}$$

and substituting Equation (5) into Equation (4) yields

$$n = \frac{4g\gamma RT_4\left(1 - \frac{T_5}{T_4}\right)}{(\gamma - 1)MU_t^2\left(\frac{r_h}{r_t} + 1\right)^2} \tag{6}$$

The previously derived equation shows that the number of stages for an axial-flow turbine depends on such factors as inlet temperature, temperature ratio, molecular weight, tip speed, and hub-to-tip radius ratio. Tip speed and, to some extent, hub-to-tip radius ratio depend on stress considerations, which will be discussed to some detail in a subsequent section dealing with the effect of temperature on the turbomachinery. Turbine temperature ratio, as previously shown, is primarily fixed by radiator area considerations. For any given inlet temperature, consequently, the number of stages is seen to be inversely proportional to fluid molecular weight. Substitution of typical parameter values into Equation (6) yields the results shown in Fig. 7 for number of stages as a function of molecular weight at a turbine inlet temperature of 2000° R. Also shown in this figure are the curves for a radial-flow turbine or compressor and an axial-flow compressor. These are obtained from consideration of the appropriate design parameters for each case. Figure 7 shows that the required number of stages is large for helium (M = 4, 5 radial turbine or compressor stages, 11 axial turbine stages, and about 50 axial compressor stages) but decreases significantly with increasing molecular weight. With neon (M = 20) the number of radial stages is reduced to 1; with argon (M = 40) the number of axial turbine stages is reduced to 1; and with krypton (M = 84) the number of axial compressor stages is reduced to 3. In general, the number of radial compressor stages equals the number of radial turbine stages, while the number of axial compressor stages is about 3 to 5 times the number of axial turbine stages.

In order to obtain a minimum number of stages, high molecular weight appears to be desirable. The heat-transfer characteristics of the inert gases, however, improve with decreasing molecular weight as pointed out in reference 3. In compromising these two opposing effects, molecular weights in the range of 20 to 40 appear attractive.

Reynolds number. - Reynolds number is directly proportional to weight flow and inversely proportional to viscosity for a given flow area. Neglecting, for the moment, the viscosity effect, Reynolds number is directly proportional to molecular weight, since reference 1 shows that weight flow is directly proportional to molecular weight. Reintroducing the viscosity effect, the relative Reynolds numbers for the fluids of interest (assuming helium as unity) are 1 for helium, 6.0 for neon, 15.7 for argon, and 27.0 for krypton.

A general effect of Reynolds number on axial-flow-compressor efficiency, based on a compilation of data (10), is presented in Fig. 8. This figure indicates that there is a reduction in achievable efficiency as Reynolds number decreases below 10^6, and this reduction becomes more pronounced with decreasing Reynolds number. The reduction in efficiency is attributed to the increased friction losses accompanying a decreasing Reynolds number. This effect has been found to be more significant for compressors than for turbines. If operation should occur at a point where the effect of Reynolds number on performance is significant, the use of a higher molecular weight gas could result in a significant improvement in performance. Reynolds number will be discussed further in a subsequent section dealing with the effect of power level on the turbomachinery.

Temperature

The effect of system temperature level on the turbomachinery is primarily a result of stress considerations in the turbine. The materials chosen for the turbine must withstand the highest temperatures and stresses in the system. Although the various materials considered must have adequate ductility and fabricability, the final selection must be based primarily on the stress-rupture and creep properties. Since space-power systems are designed for long-time operation (10,000 hr), the selection of a suitable turbine material, especially for temperatures in excess of 2000° R, is a major consideration. Rupture and creep properties of most materials have been obtained only for relatively short times, on the order of 100 to 500 hr. Data for the newer alloys are even based on less time. It is, therefore, necessary to extrapolate these data to 10,000 hr. Such an extrapolation over two or more orders of magnitude of time results in some degree of uncertainty in the extrapolated values.

Extrapolation of the short-time stress-rupture data was accomplished by using the Larson-Miller parameter, an empirical parameter commonly used for such an extrapolation. A survey of materials for use at turbine inlet temperatures of 2000° to 3000° R showed that the high-strength alloys of nickel, molybdenum, and tungsten have the most attractive characteristics for covering this temperature range. Rupture stresses extrapolated to 10,000 hr are shown as a function of temperature in Fig. 9 for typical alloys of the previous types. The nickel alloy is usable at the lower end of the temperature range (around the 2000° R level) while the molybdenum alloy has sufficient strength for possible use up to about 2700° or 2800° R. For higher temperatures, the tungsten alloy must be used. The nickel and molybdenum alloys typified in Fig. 9 are commercially available, while the tungsten alloy is still in the research stage. For this study, the allowable stress was chosen as the stress resulting in 1-percent creep in 10,000 hr of operation. This allowable stress was estimated by using 50 percent of the extrapolated rupture stress. This is a standard estimation procedure that usually results in a conservative stress value.

As was previously shown in Equation (6), the required number of stages for an axial-flow turbine is directly proportional to the turbine inlet temperature, inversely proportional to the blade-tip velocity squared, and an inverse function of the hub-to-tip radius ratio. It will now be shown that a hub-to-tip radius ratio can be obtained from stress considerations, and the allowable tip speed is a function of the allowable stress and material density. An equation relating blade and disk parameters, for a constant-stress disk, is derived in reference 11 and can be rearranged to yield

$$\ln \frac{y_a}{y_r} = \frac{\rho_d s_b (r_r/r_h)^2}{\rho_b s_d \left[(r_t/r_h)^2 - 1\right]} \tag{7}$$

where y_a/y_r is the ratio of disk thickness at the axis to disk thickness just below the rim. With the assumption of typical values of $y_a/y_r = 3$ and $r_r/r_h = 0.95$ and letting $\rho_b = \rho_d$, Equation (7) can be solved for r_h/r_t:

$$\frac{r_h}{r_t} = \left(1 + 0.821 \frac{s_b}{s_d}\right)^{-1/2} \tag{8}$$

Since blade bending stresses are not analyzed in this study, s_b/s_d is assumed equal to 0.6 in order to allow margin for these, and Equation (8) yields $r_h/r_t = 0.818$, which is a reasonable value for the hub-to-tip radius ratio.

An equation for blade centrifugal stress is derived in reference 11 as

$$s_b = \frac{\rho_b U_t^2}{288g}\left[1 - \left(\frac{r_h}{r_t}\right)^2\right] \tag{9}$$

If $\rho_b = \rho_d$, $s_b = 0.6\ s_d$, and $r_h/r_t = 0.818$, Equation (9), when solved for U_t^2, yields

$$U_t^2 = 1.609 \times 10^4 \frac{s_d}{\rho_d} \tag{10}$$

Substituting Equation (10) into the previously derived equation (Eq. (6)) for number of axial stages and setting $T_5/T_4 = 0.8$, M = 40 (argon), and $r_h/r_t = 0.818$ yields, after evaluation of the numerical constants,

$$n = 0.0448 \frac{T_4}{(s_d/\rho_d)} \tag{11}$$

For any given fluid, as seen from Equation (11), the required number of stages is directly proportional to the turbine inlet temperature and inversely proportional to the material stress to density ratio. Since the turbine rotor is not exposed to the turbine inlet temperature, the allowable stress used in Equation (11) was taken at a temperature 100° below the inlet temperature. Experience has shown this to be a reasonable estimate. A similar type of relationship can be derived for a radial-flow turbine by starting with the stress equation presented by Wood (6).

The required number of turbine stages as a function of inlet temperature is presented in Fig. 10 for both axial- and radial-flow turbines each with a molybdenum alloy and a tungsten alloy rotor. As turbine inlet temperature increases, the number of stages increases rapidly. For the case of a molybdenum-alloy rotor the required number of stages for an axial-flow turbine increases from 1 to 14 as turbine inlet temperature increases from 2000° to 2800° R, while for a radial-flow turbine the required number of stages increases from 1 to 6 as turbine inlet temperature increases from 2200° to 2800° R. For a tungsten-alloy rotor the number of stages is about one third that required with a molybdenum alloy. Although the allowable stress for the tungsten alloy, as seen from Fig. 9, is about 6 times that for the molybdenum alloy, the density of tungsten is about 2 times that of molybdenum, thus resulting in a threefold improvement in stress to density ratio for tungsten over molybdenum. As was mentioned previously, the number of radial-flow-compressor stages is equal to the number of radial-flow-turbine stages, while the number of axial-flow-compressor stages is about 3 to 5 times the number of axial-flow-turbine stages.

Pressure and Power

The effects of pressure level and power level on the turbomachinery are discussed together since, as will be shown, both their choice and the manner in which they affect the rotating components are closely related. The pressure and the power levels affect the turbomachinery primarily in the areas of size (diameter) and Reynolds number.

Diameter. - For any one of the turbomachines under consideration, the diameter is determined primarily from continuity considerations; that is, a definite flow area is needed in order to pass the given flow. As an example, an axial-flow turbine will be considered. At the last-stage-rotor exit the required flow area, as defined by continuity, is

$$A = \frac{w}{\rho_{ex}V_{x,ex}} = \frac{wRT_{ex}}{p_{ex}MV_{x,ex}} \tag{12}$$

This area is the annulus area between blade hub and tip and is equal to

$$A = \pi\left(r_t^2 - r_h^2\right) = \frac{\pi d_t^2}{4}\left[1 - \left(\frac{r_h}{r_t}\right)^2\right] \tag{13}$$

Equating these two expressions for area and solving for tip diameter yields

$$d_t = \left\{\frac{4wRT_{ex}}{\pi\left[1 - \left(\frac{r_h}{r_t}\right)^2\right]p_{ex}MV_{x,ex}}\right\}^{1/2} \tag{14}$$

Weight flow, as shown in reference 1, is directly proportional to molecular weight and power and inversely proportional to temperature level. Assuming exit velocity and hub-to-tip radius ratio constant, it is seen that

$$d_t \propto \left(\frac{\text{power}}{\text{pressure}}\right)^{1/2}$$

This relationship holds true for all the turbomachines, turbine or compressor, axial or radial.

Typical curves for the diameter of an axial-flow turbine as a function of pressure are presented in Fig. 11 for system power levels of 10, 100, and 1000 kw. For any given power level, the diameter decreases with increasing pressure according to the inverse square root relationship shown previously. For a 10 kw system, an increase in turbine inlet pressure from 5 to 30 psia results in a decrease in diameter from 10 to 4 in.. For other power levels the pressures corresponding to similar diameters are in direct proportion to the power.

The size of a turbine or compressor appears to affect the achievable efficiency level. With decreasing diameter, factors such as tip and running clearance losses have a more pronounced effect on overall efficiency. The exact magnitude and extent of this size effect is not well defined. For the achievement of high efficiency, however, relatively large size turbomachinery appears to be desirable. Pressure level, consequently, must be chosen so that the turbomachinery does not become too small to achieve high efficiency. Since the heat-transfer components benefit from relatively high pressures, as pointed out in reference 3, the selection of a system pressure level must be made on the basis of a compromise.

As power level increases, the pressure level corresponding to a given turbomachinery diameter increases in direct proportion. Consequently, turbomachinery with a given geometry can be used in systems of different power outputs by appropriately adjusting the pressure level. As power output increases, the allowable system pressure can increase and, as pointed out previously, this benefits the heat-transfer components. For a given set of turbomachinery, the amount of pressure increase, of course, is governed by the maximum allowable design stress of the component casing and by the bearing-load capability due to the higher axial-thrust loads on the rotors. Even with a redesign of the casing and running gear, pressure level cannot be increased indefinitely with power level because a limiting pressure level, as dictated by system structural considerations, is eventually reached. Studies have indicated that the limiting pressure level is reached at approximately the 200- to 400-kw power level. At power levels above this, pressure can be maintained constant and turbomachinery diameter will increase as the square root of power.

Reynolds number. - As indicated previously, a significant reduction in compressor efficiency may occur as Reynolds number decreases well below 10^6. There are several ways to express Reynolds number for a turbomachine, but they are all of the general form

$$N_R \propto \frac{w}{\mu d}$$

Expressing weight flow and diameter in terms of the parameters that affect them yields

$$N_R \propto \frac{PM}{\mu T(P/p)^{1/2}}$$

For a given temperature and fluid, it is seen that (a) in the range of power outputs (below about 300 kw) where pressure and power are maintained in direct proportion, Reynolds number increases directly as power, and (b) at higher power outputs where pressure is maintained constant, Reynolds number increases as the square root of power. With argon as the working fluid, typical Reynolds numbers range from about 10^6 at the 1000-kw power level to less than 10^5 at the 10-kw power level. For neon, the corresponding Reynolds numbers are reduced by a factor of about 2.5. At the low power levels, consequently, compressor performance may very well be affected by Reynolds number considerations.

TURBOMACHINERY SELECTION

The selection of turbomachinery for any system design involves many considerations. In addition to the previously discussed system design parameters, consideration must be given to such areas as (a) the choice of axial- or radial-flow machines and (b) the use of a single turbine or separate turbines for driving the compressor and the alternator. These items are discussed in this section and typical turbomachinery configurations are presented for systems having power outputs of 10, 100, and 1000 kw.

Turbomachinery Type

The decision whether to use radial-flow or axial-flow turbomachinery for any given system is usually based on (a) the required number of stages and (b) the rotor tip diameter. Where multistaging is required, a purely radial-flow configuration appears to be less attractive than other possible configurations. The required ducting and associated duct losses for multistage radial-flow configurations reduce their potential for achieving maximum efficiencies. For situations requiring multistaging, therefore, the best choice appears to be a purely axial-flow machine or, in cases requiring a large number of stages, a hybrid configuration (1 radial stage plus axial stages).

The use of a single-stage radial-flow configuration appears to be of the most advantage where small size machines are required. In the region of small tip diameters, about 5 in. and less, the performance of radial-flow machines is less sensitive to flow surface deviations than is the performance of axial-flow machines. The small throat areas required for small-diameter axial-flow machines present problems in fabrication because of the required close tolerances. The amount of tip clearance must be held as small as possible since this clearance has a significant effect on the performance of axial-flow machines, especially those with small blade heights. For small size radial-flow machines, clearances are not as critical. The single-stage radial-flow machine, therefore, appears most suitable for use in the small-size region since performance is less sensitive to size effects.

Turbomachinery Arrangement

There are two possible arrangements for the turbomachinery components and these are shown in Fig. 12. Figure 12(a) shows a single turbine that is directly coupled to the compressor and alternator, while Fig. 12(b) shows a dual turbine arrangement whereby one turbine drives the compressor and the second turbine drives the alternator. The dual turbine arrangement appears to be attractive for situations where alternator requirements dictate comparatively low rotative speeds while compressor requirements dictate high rotative speeds. If frequency requirements are 400 cycles per second, for example, the required speed for the alternator would be 24,000 rpm at maximum and very possibly only 12,000 rpm. Such a low rotative speed could result in an excessive number of stages for the compressor. A dual turbine system would allow both the alternator and the compressor to operate at optimum speeds without the need for a speed reduction gearbox.

Another factor in favor of the dual turbine arrangement is that it results in lower turbine-exit kinetic-energy losses when compared to the single turbine arrangement. Typical exit losses for the single turbine arrangement (without a diffuser) are approximately 6 to 10 percent of turbine work, while exit losses for the dual turbine arrangement are about 2 to 3 percent of the total turbine work. The low-speed alternator turbine, therefore, serves as an efficient diffuser.

Typical Turbomachinery Designs

Application of the principles and the considerations discussed in this paper will result in the preliminary specification of turbomachinery features capable of yielding highly efficient and reliable operation. In order to show typical Brayton cycle turbomachinery configurations, three systems with different power and temperature levels were chosen as example cases. These three cases were subjected to more detailed analyses than those discussed previously; the resultant geometry features, consequently, will vary somewhat from those shown in the trend curves. The power, temperature, and selected pressure levels for these three cases are as follows:

Power output, kw	Turbine inlet temperature, °R	Compressor inlet pressure, psia
10	1950	6
100	2300	75
1000	3000	200

Argon was chosen as the working fluid for all cases, and an alternator speed of 12,000 rpm was set as a requirement. The turbomachinery geometry features and design parameters are presented for each case.

10-Kilowatt system. - For this comparatively low-temperature (1950° R turbine inlet) system, a high-strength nickel alloy can be chosen as the turbine rotor material. A dual turbine configuration was selected because of the large difference in speed requirements between the compressor and the alternator. An overall turbine efficiency of 85 percent (total-to-static) and a compressor efficiency of 80 percent were chosen as design goals.

The pertinent characteristics of the compressor and compressor-drive turbine are presented in Fig. 13(a). Both radial- and axial-flow configurations are presented. A single radial-flow stage for both the compressor and the turbine is suitable for this application, since the specific speeds and pressure ratios are favorable to the attainment of the required efficiencies. This unit would rotate at a speed of 42,000 rpm and the tip diameters for the compressor and turbine are 5.38 and 5.51 in., respectively. A 6-stage axial-flow compressor and a single-stage axial-flow turbine are also suitable for this application. This unit would rotate at 44,700 rpm and have tip diameters of 4.25 (inlet) and 5.50 in. for the compressor and turbine, respectively.

The pertinent characteristics for the alternator-drive turbine, for both radial- and axial-flow configurations, are shown in Fig. 13(b). For the required 12,000 rpm, the radial-flow turbine is a single-stage unit having a tip diameter of 12.90 in., while the axial-flow turbine is a two-stage unit having a tip diameter of 9.34 in.. The specific speed for the radial-flow turbine is comparatively low (57 as compared to the optimum range of 80 to 100), and the axial-flow turbine probably would be preferable for this application.

100-Kilowatt system. - At this intermediate temperature level (2300° R turbine inlet), a high-strength molybdenum alloy was chosen for the turbine. The temperature is too high for obtaining an attractive configuration with a nickel alloy. Here again a dual turbine configuration was selected because of the low alternator speed. Design efficiency goals of 85 and 80 percent were specified for the turbine and compressor, respectively.

The pertinent characteristics for the compressor and compressor-drive turbine are presented in Fig. 14(a). A single-stage radial-flow configuration appeared most attractive for both the compressor and the turbine. The rotative speed is 55,900 rpm, and the tip diameters for the compressor and turbine are 4.64 and 4.76 in., respectively. Axial-flow turbomachinery for this application would require about 2 stages for the turbine and about 5 stages for the compressor.

The pertinent characteristics for the alternator-drive turbine are presented in Fig. 14(b) for both an axial-flow and a hybrid configuration. The 6-stage axial-flow turbine has a tip diameter of 6.50 in.. A hybrid configuration results in the total number of stages being reduced to 4 (1 radial plus 3 axial). The radial stage has an 11.27-in. tip diameter and the axial stages have a 6.54-in. tip diameter. Either configuration should be satisfactory for this application.

1000-Kilowatt system. - For this sytem, the turbine presents a major problem due to the high turbine inlet temperature of 3000° R. This high temperature application could very possibly derive much benefit from the use of turbine cooling. Due to the preliminary nature of the analysis, however, turbine cooling and the associated advantages and disadvantages were not considered at this time. At this high temperature level, the choice of materials for the turbine, and especially for the highly stressed blades and disk, becomes quite critical and quite limited. Only the highest strength refractory-metal alloys can be considered for use.

Consideration was first given to the dual turbine concept. For a preliminary stress analysis, three configurations were selected for the "hot" compressor-drive turbine: a single-stage radial and two three-stage hybrid turbines. The stress analysis indicated that a radial single stage or a hybrid turbine was not feasible because the stress levels of the radial stage, for all cases, were too high even for a high-strength tungsten alloy as structural material.

From the combined standpoints of stress and performance, axial-stage turbomachinery appeared to be the logical choice. Since the rotative speed must be low in order to stay within the stress limits, a single multistage turbine was chosen for driving both the compressor and the alternator. The pertinent characteristics of this turbine and the associated compressor are presented in Fig. 15. A five-stage axial-flow turbine constructed of a high-strength tungsten alloy and rotating at 12,000 rpm was found to be well within the stress limit. The tip diameter of this turbine was 9.16 in. at the inlet and increased to 13.5 in. at the exit. A varying mean-diameter design was chosen in order to permit higher blade speeds in the latter stages where lower temperatures and higher allowable stresses exist. This type of configuration results in a fewer number of stages than a constant mean-diameter configuration, which would require 8 stages.

The recommended compressor was a constant-diameter axial-flow machine with 12 stages and a 12.5-in. tip diameter. A hybrid compressor was also considered, but it did not offer any significant reduction in the total number of stages.

CONCLUDING REMARKS

This study of Brayton cycle turbomachinery characteristics was part of an overall investigation of Brayton cycle systems undertaken at the Lewis Research Center. Basic turbomachinery requirements of high efficiency and reliability are shown to be dictated by system considerations. The requirement for high efficiency is met basically through the use of optimum values for turbomachinery design parameters, such as speed-work parameter, diffusion factor, and specific speed. Turbomachinery reliability for intended missions is achieved primarily through adherence to stress and creep limitations.

System design parameters having a major effect on turbomachinery geometry are fluid molecular weight, temperature, pressure, and power. Increasing fluid molecular weight and decreasing system temperature level result in a reduction in the required number of turbomachinery stages. Pressure must be low enough to yield turbomachinery sizes that are consistent with good performance. Since other system requirements are more favorably satisfied with low molecular weight fluids, high temperatures, and high pressures, compromises must be made in the selection of these parameters. Increasing power level allows the use of higher pressure without compromising turbomachinery size and performance.

Typical turbomachinery designs are presented for systems with power outputs of 10, 100, and 1000 kw in order to show the results of applying the principles discussed in the paper.

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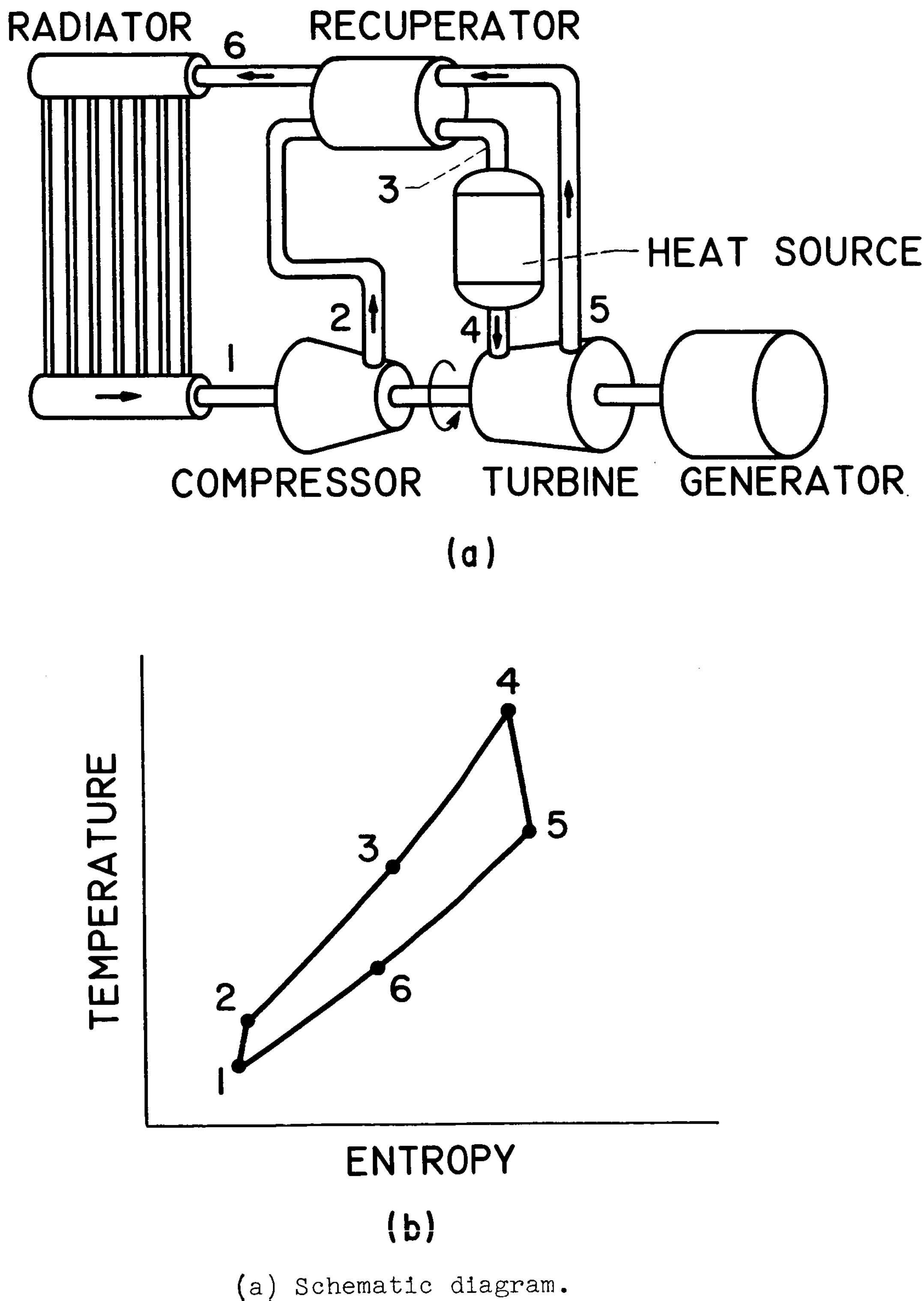

(a) Schematic diagram.
(b) Temperature-entropy diagram.

Figure 1. - Brayton power cycle.

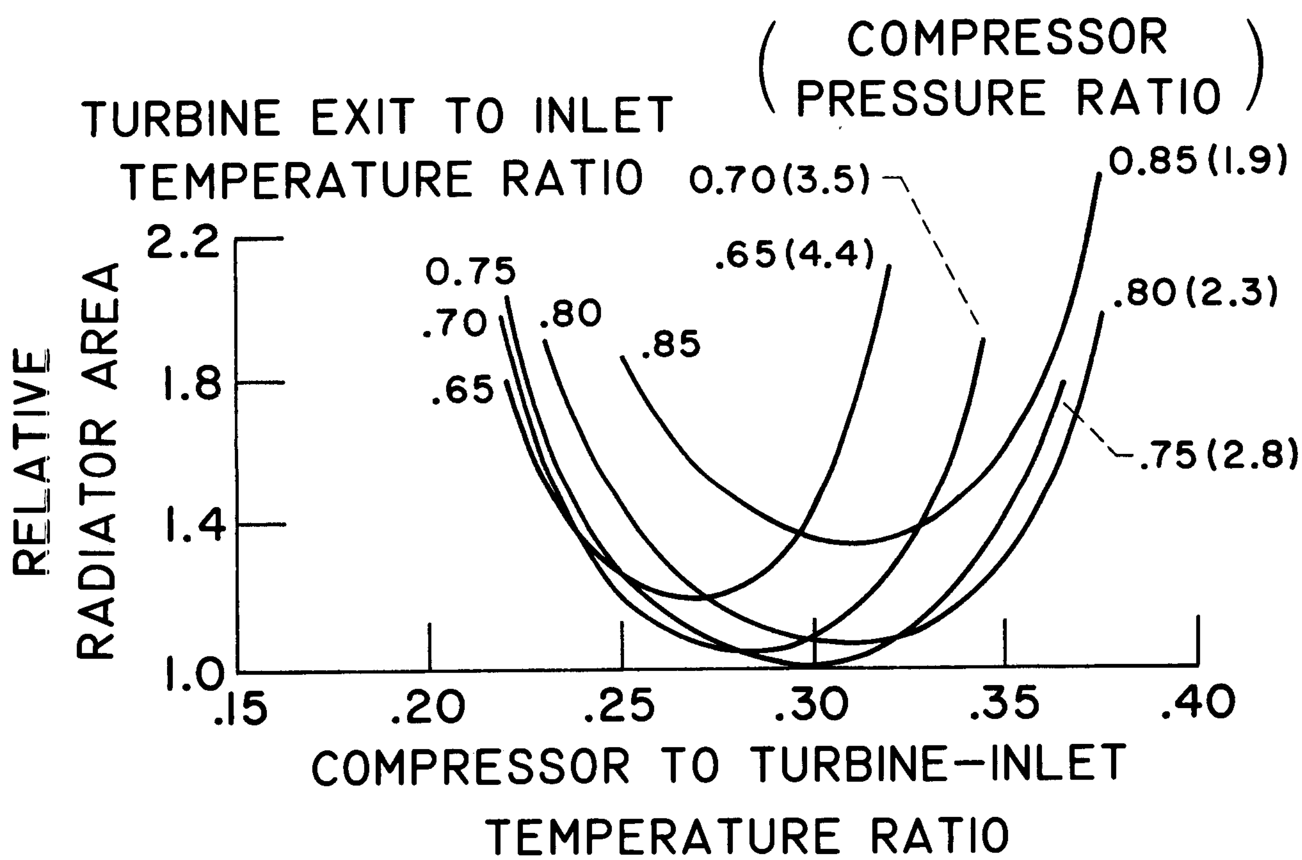

Figure 2. - Effect of cycle temperature variables on radiator area.

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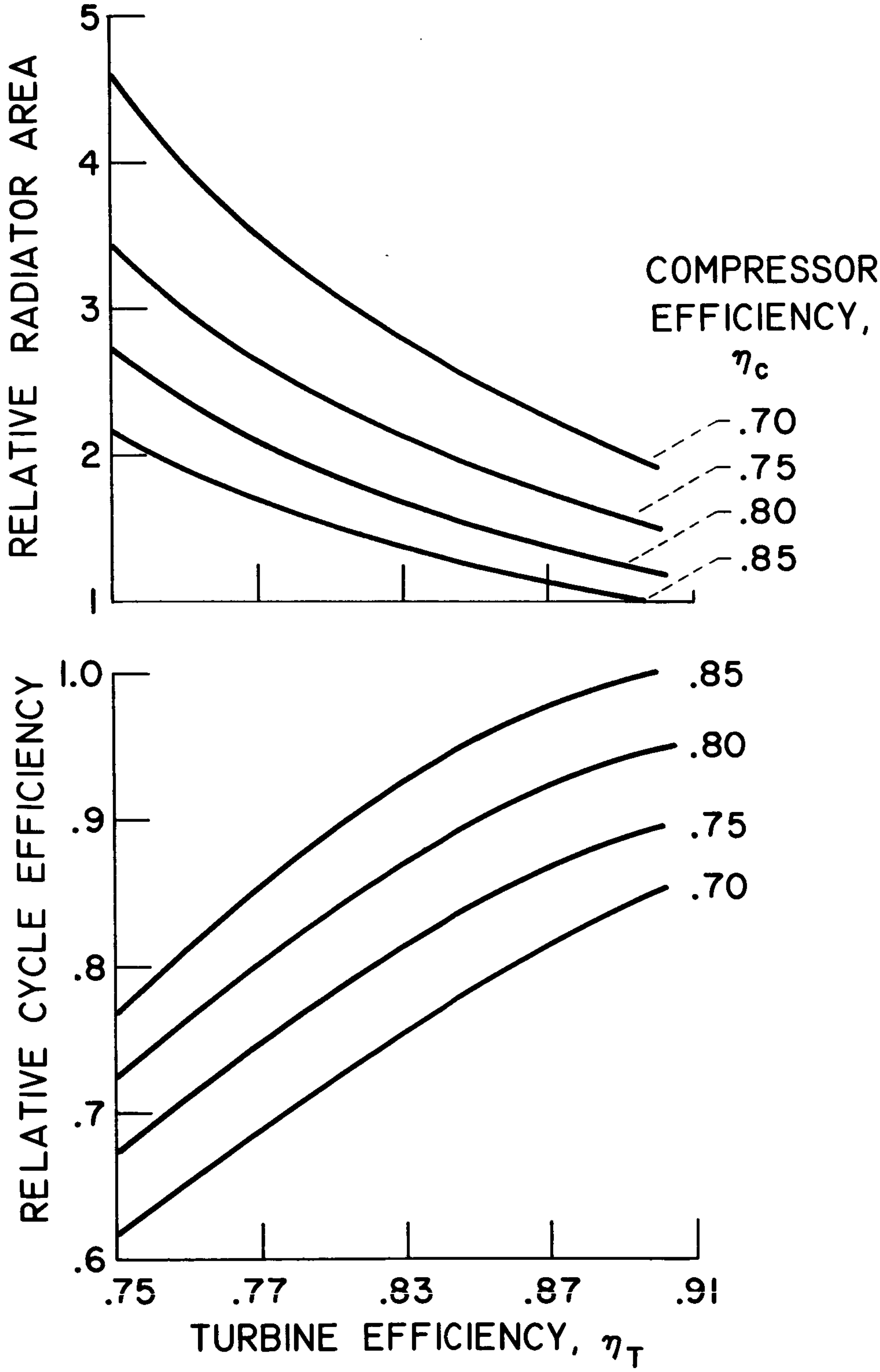

Figure 3. - Effect of turbomachinery efficiency on radiator area and cycle efficiency (for minimum area operating points).

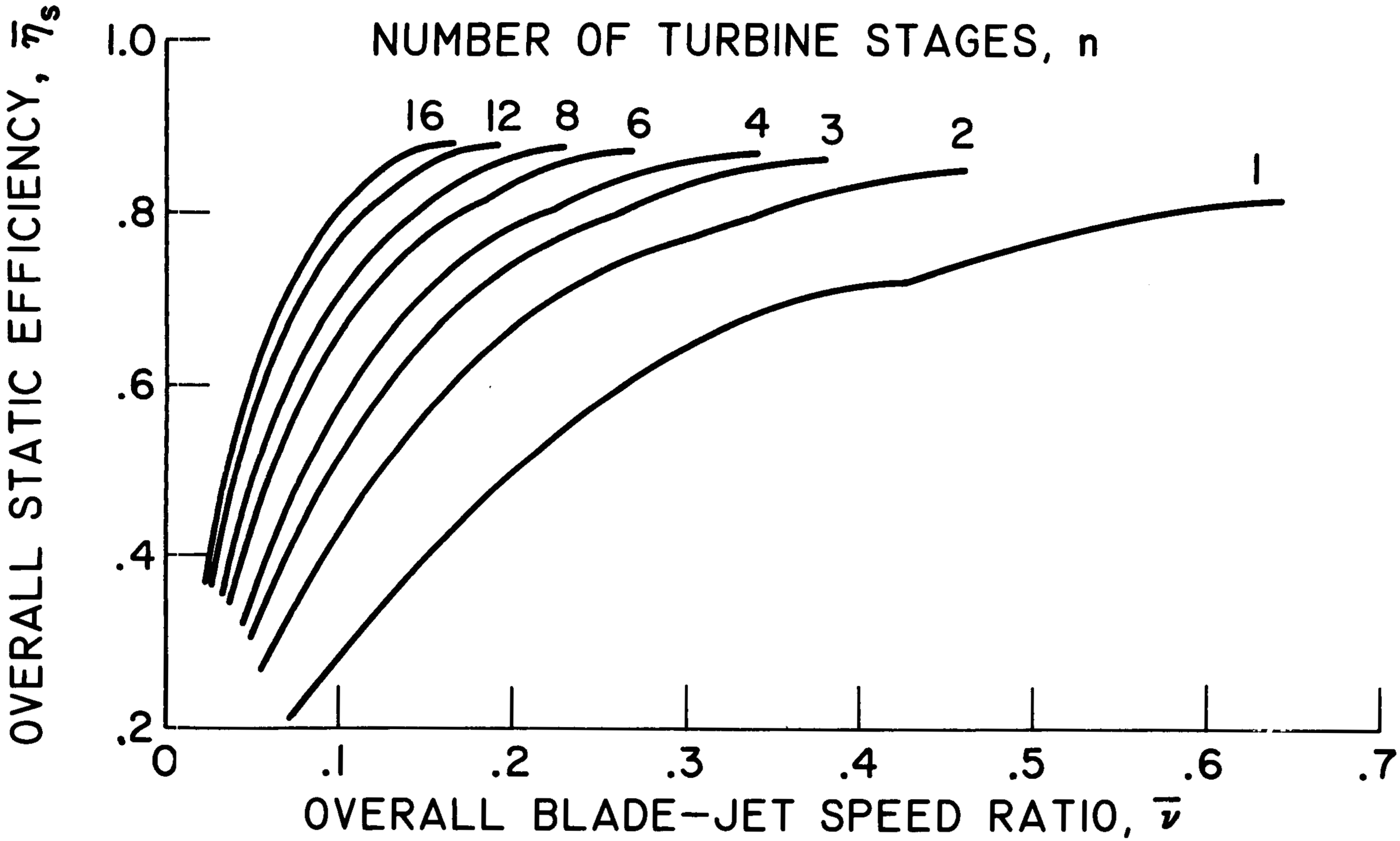

Figure 4. - Multistage axial-flow turbine static-efficiency characteristics as a function of overall blade-jet speed ratio.

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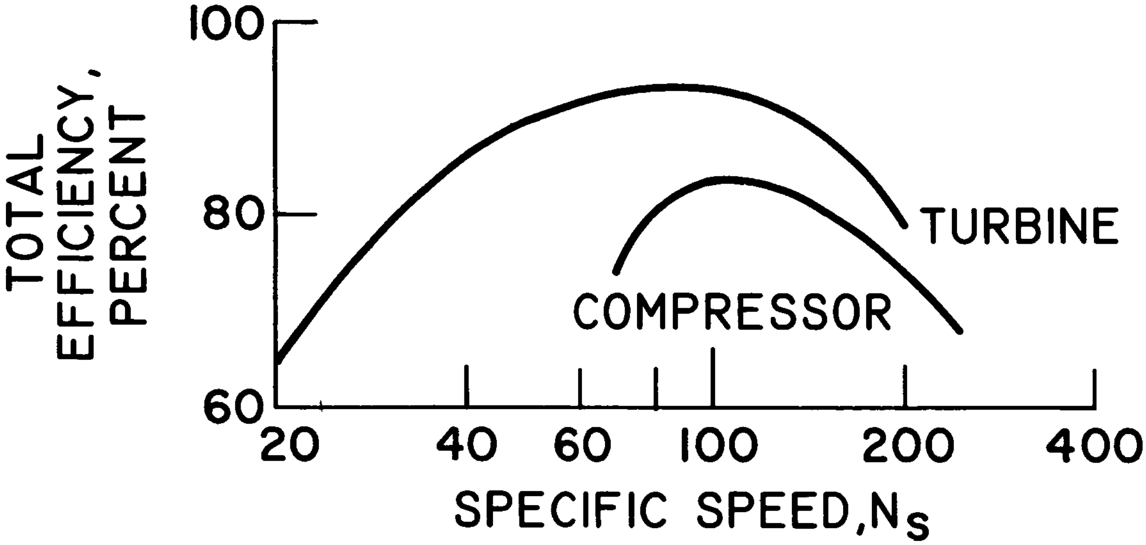

Figure 5. - Variation of total efficiency with specific speed for radial-flow turbomachinery.

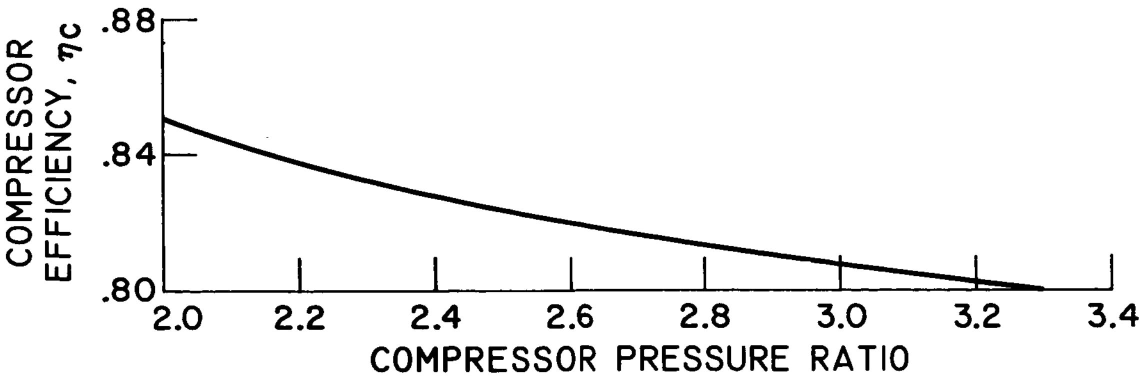

Figure 6. - Typical effect of stage pressure ratio on radial-flow compressor efficiency.

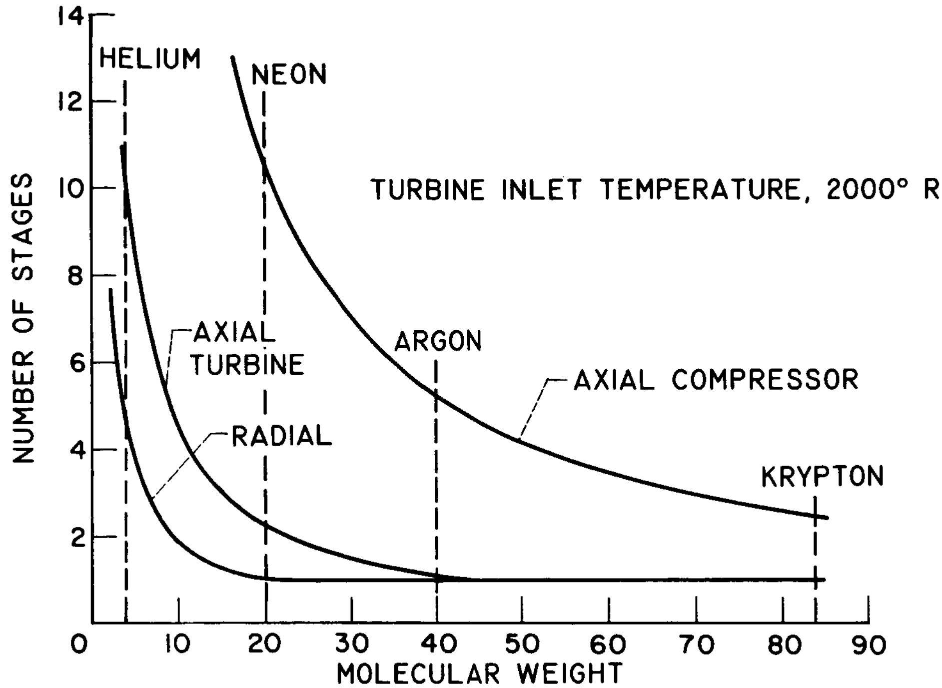

Figure 7. - Effect of molecular weight on number of stages.

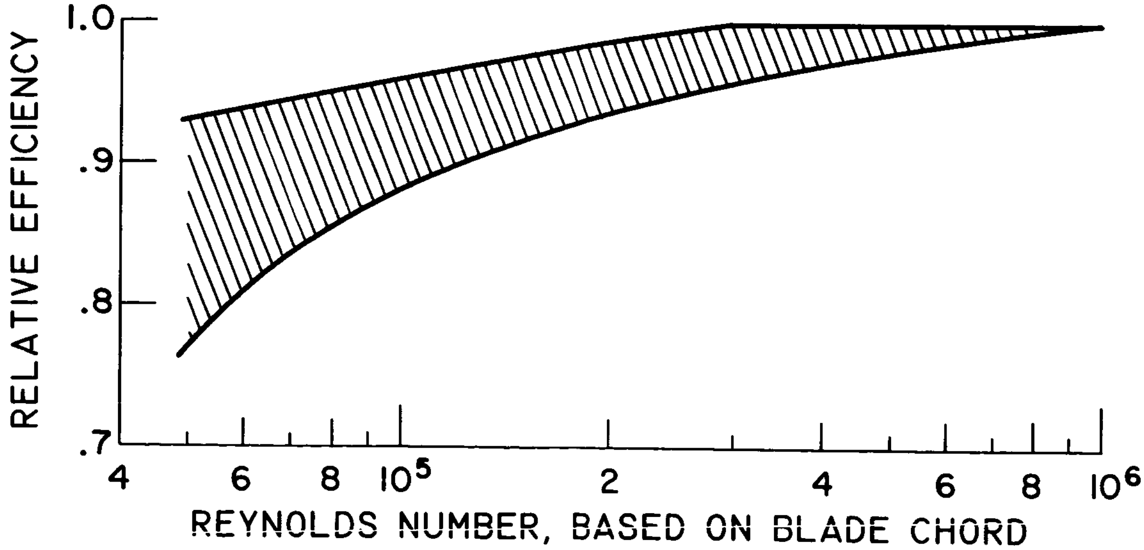

Figure 8. - Effect of Reynolds number on axial-flow compressor efficiency.

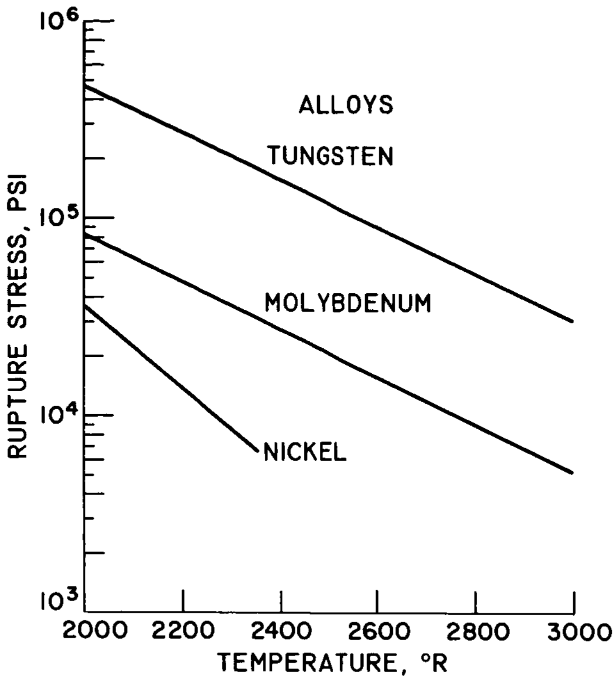

Figure 9. - Stress-rupture curve at 10,000 hrs for high-strength alloys.

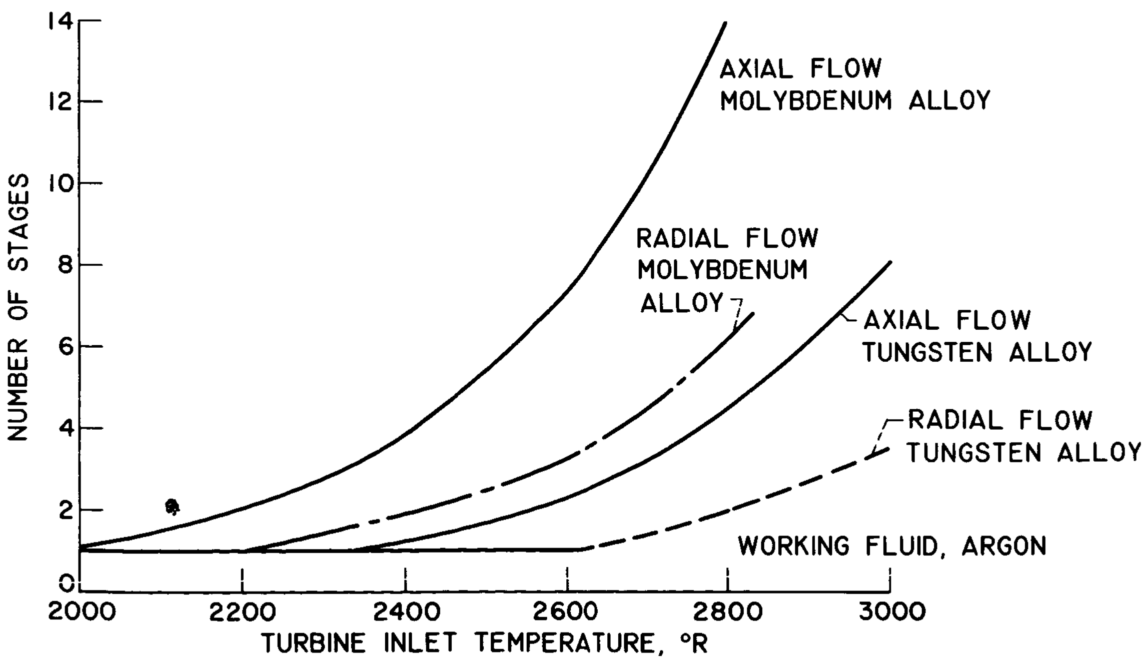

Figure 10. - Effect of turbine inlet temperature on number of turbine stages.

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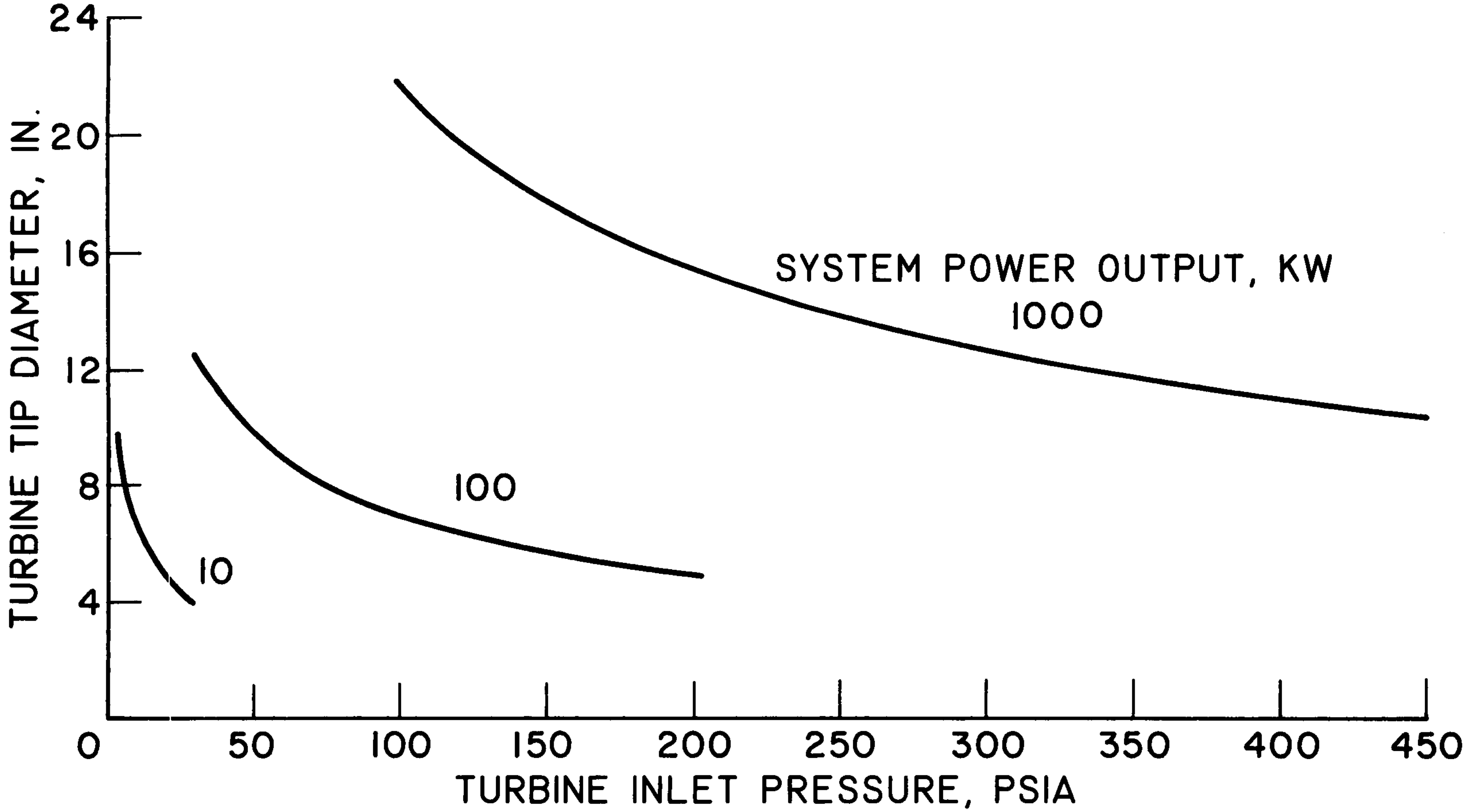

Figure 11. - Effect of inlet pressure on axial-flow turbine diameter.

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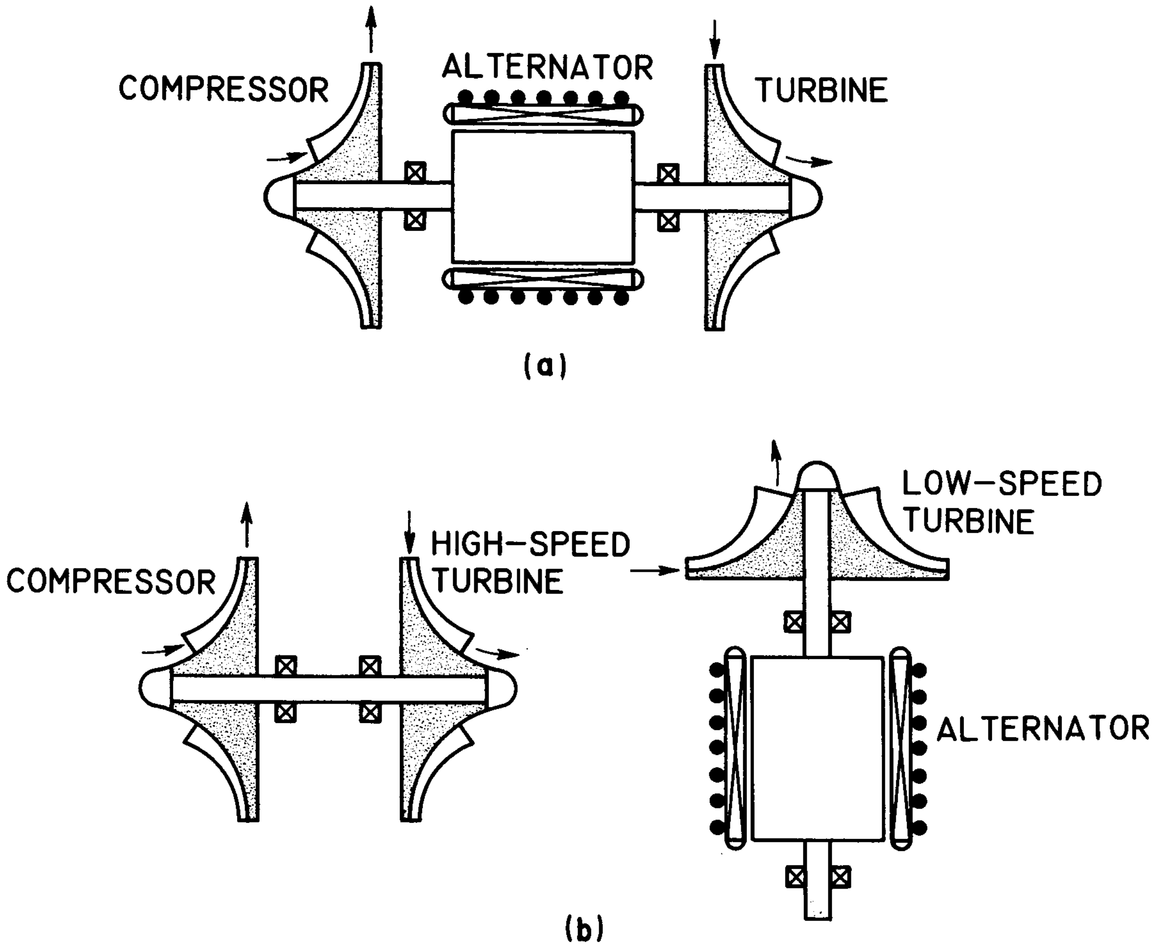


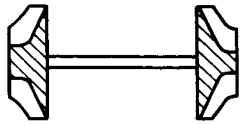
(a) Single turbine.
(b) Dual turbine.

Figure 12. - Turbomachinery arrangements.

RADIAL-FLOW CONFIGURATION

COMPRESSOR	
INLET TEMP.	536^{0} R
INLET PRESSURE	6.0 psia
PRESSURE RATIO	2.30
SPECIFIC SPEED	95
STAGES	1
TIP DIAMETER	5.38 in.
RPM	42,000

COMPRESSOR TURBINE

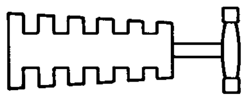


TURBINE	
INLET TEMP.	1950^{0} R
INLET PRESSURE	13.2 psia
PRESSURE RATIO	1.56
SPECIFIC SPEED	104.5
STAGES	1
TIP DIAMETER	5.51 in.
RPM	42,000
MATERIAL	NICKEL ALLOY

AXIAL-FLOW CONFIGURATION

COMPRESSOR	
INLET TEMP.	536^{0} R
INLET PRESSURE	6.0 psia
PRESSURE RATIO	2.30
DIFFUSION (TIP)	0.35
STAGES	6
TIP DIAMETER (INLET)	4.25 in.
RPM	44,700

COMPRESSOR TURBINE



TURBINE	
INLET TEMP.	1950^{0} R
INLET PRESSURE	13.2 psia
PRESSURE RATIO	1.56
SPEED-WORK PARAMETER	1.0
STAGES	1
TIP DIAMETER	5.50 in.
RPM	44,700
MATERIAL	NICKEL ALLOY

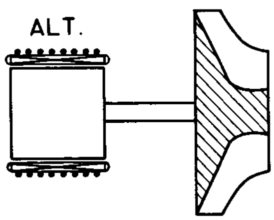
(a) Compressor and compressor-drive turbine.

Figure 13. - Turbomachinery for 10-kilowatt Brayton power system.

RADIAL-FLOW CONFIGURATION

TURBINE

ALT.

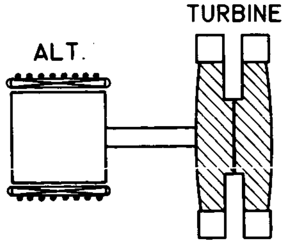


INLET TEMP.	1685^{0} R
INLET PRESSURE	8.45 psia
PRESSURE RATIO	1.26
SPECIFIC SPEED	57
STAGES	1
TIP DIAMETER	12.90 in.
RPM	12,000

AXIAL-FLOW CONFIGURATION

TURBINE

ALT.



INLET TEMP.	1685^{0} R
INLET PRESSURE	8.45 psia
PRESSURE RATIO	1.26
STAGE SPEED-WORK PARAMETER	1.00
STAGES	2
TIP DIAMETER	9.34 in.
RPM	12,000

(b) Alternator-drive turbine.

Figure 13. - Concluded. Turbomachinery for 10-kilowatt Brayton power system.

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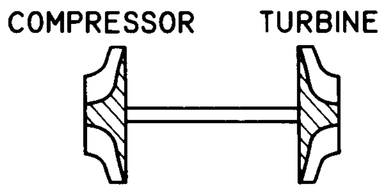
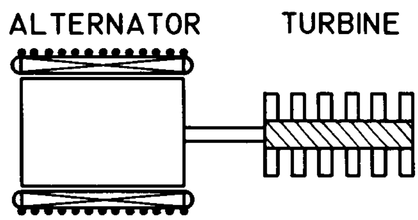
INLET TEMP.	632^{0} R
INLET PRESSURE	75 psia
PRESSURE RATIO	2.5
SPECIFIC SPEED	89
STAGES	1
TIP DIAMETER	4.64 in.
RPM	55,900



INLET TEMP.	2300^{0} R
INLET PRESSURE	182.5 psia
PRESSURE RATIO	1.64
SPECIFIC SPEED	98
STAGES	1
TIP DIAMETER	4.76 in.
RPM	55,900
MATERIAL	MOLYBDENUM ALLOY

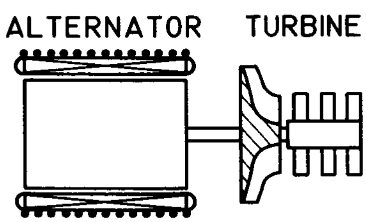
(a)

AXIAL–FLOW CONFIGURATION



INLET TEMP.	1953^{0} R
INLET PRESSURE	111.0 psia
PRESSURE RATIO	1.34
STAGE SPEED-WORK PARAMETER	1.0
STAGES	6
TIP DIAMETER	6.50 in.
RPM	12,000

HYBRID CONFIGURATION

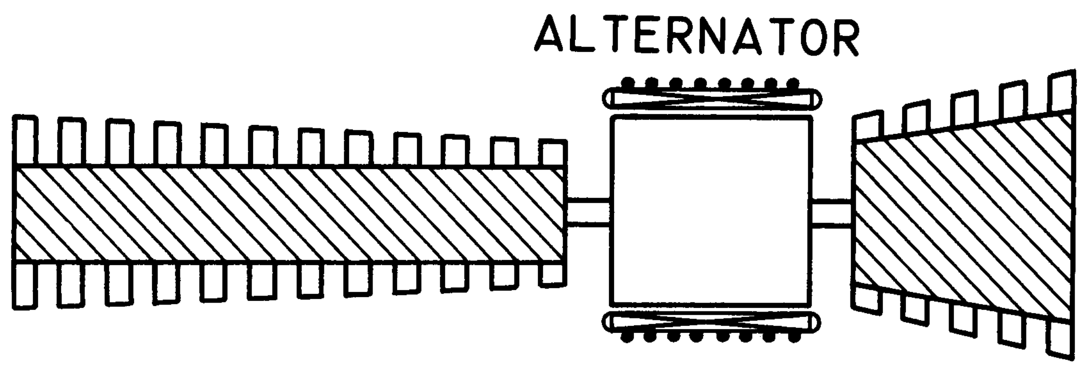


INLET TEMP.	1953^{0} R
INLET PRESSURE	111.0 psia
PRESSURE RATIO	1.34
SPECIFIC SPEED (RADIAL)	58
SPEED-WORK PARAMETER (AXIAL)	1.0
STAGES (TOTAL)	4
TIP DIAMETER (RADIAL)	11.27 in.
TIP DIAMETER (AXIAL)	6.54 in.
RPM	12,000

(b)

(a) Compressor and compressor-drive turbine.
(b) Alternator-drive turbine.

Figure 14. - Turbomachinery for 100-kilowatt Brayton power system.



COMPRESSOR	
INLET TEMP.	975^{0} R
INLET PRESSURE	200 psia
PRESSURE RATIO	2.10
DIFFUSION (TIP)	0.40
STAGES	12
TIP DIAMETER	12.5 in.
RPM	12,000

TURBINE	
INLET TEMP.	3000^{0} R
INLET PRESSURE	408 psia
PRESSURE RATIO	1.89
STAGE SPEED-WORK PARAMETER	0.80
STAGES	5
TIP DIAMETER (INLET)	9.16 in.
TIP DIAMETER (EXIT)	13.50 in.
RPM	12,000
MATERIAL	TUNGSTEN ALLOY

Figure 15. - Turbomachinery for 1000-kilowatt Brayton power system.